Potentials of Speed and Displacement Variable Pumps in Hydraulic Applications

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Abstract

Speed and displacement variable pumps offer a degree of freedom for process control. As a certain operation point can be supplied by different combinations of drive speed and pump displacement intelligent control strategies can address major issues like energy efficiency, process dynamics and noise level in industrial applications. This paper will provide an overview of recent research and development activities to evaluate the named potentials.

KEYWORDS: Energy efficiency, performance, speed variable pump, industrial hydraulics

1. Introduction

Nowadays numerous approaches for controlling hydraulic processes are available for machine manufacturers. In addition to the conventional valve control various displacement controlled concepts have been established due to a rising demand of energy efficient drive systems /1/. Besides variable displacement pumps operated at a constant speed level, falling prices of frequency converters match the awareness of energy efficiency and lead to an increased use of speed variable pump drives in recent years /3, 4, 6/. Combining these two concepts results in a speed and displacement variable pump that can be interpreted as a hydraulic gearbox between the electric drive and the actual process. These so called HydroGear systems allow to leverage the advantages of both control principles while eliminating known drawbacks at the same time /5/. As such, low drive speeds can increase the energy efficiency especially at partial load or process-related idle time. A variable pump displacement can additionally reduce the motor torque in pressure holding sequences. However, to fully benefit from the degree of freedom efficient control strategies are necessary and will be discussed in the following chapters.

2. Improving Energy Efficiency with HydroGear Systems

The degree of freedom of speed and displacement variable pumps can be used to adjust the operation points of the electric drive and the hydraulic pump to maximize the overall energy efficiency. Depending on the process dynamics there are mainly two different concepts.

2.1. Conventional Energy Optimization

For a given hydraulic system pressure and a demanded volume flow, the drive speed and the pump displacement will be set to minimize the overall losses of the electro hydraulic drive system.

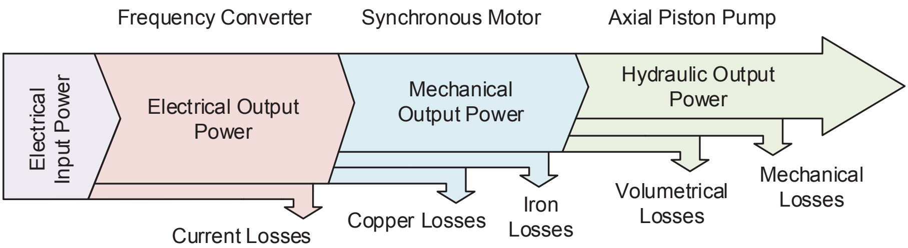

Figure 1: Dominant system losses of HydroGear systems

As shown in **Figure 1** the objective is to minimize the electric input power for a given hydraulic output power. To reduce the complexity of this optimization task only the dominant losses of each component are considered /2, 7/:

Frequency converter:

- Current losses $P_{FC} = K_{FC} \cdot |I|$ (1)

Electric drive:

- Copper losses $P_{Cu} = 3 \cdot R_{Cu} \cdot I^2$ (2)
- Iron losses $P_{Fe} = K_{f,1} \cdot n + K_{f,2} \cdot n^2$ (3)

Hydraulic pump:

- Volumetric losses $P_{Vol} = Q_L \cdot p$, with $Q_L = Q_L(p, Q, n)$ (4)

- Mechanical losses $P_{Mech} = M_R \cdot n$, with $M_R = M_R(p, Q, n)$ (5)

By summarizing all terms the maximization of the energy efficiency can be transformed into a mathematical optimization problem. In /5/, therefore, all partial losses are formulated in dependency of the drive speed which takes the role of a decision variable.

$$\min\{P_{tot}\} = \min\left\{P_{FC} + P_{Cu} + P_{Fe} + P_{Vol} + P_{Mech}\right\} = \min\left\{P_{tot}(\boldsymbol{n})\right\} \quad (6)$$

The solution of this objective function is the optimum drive speed to serve the given hydraulic operation point. **Figure 2** shows exemplary the optimized speed level n_{opt} as well as the corresponding pump displacement α_{opt} for a selected HydroGear system for varying system pressure and volume flow.

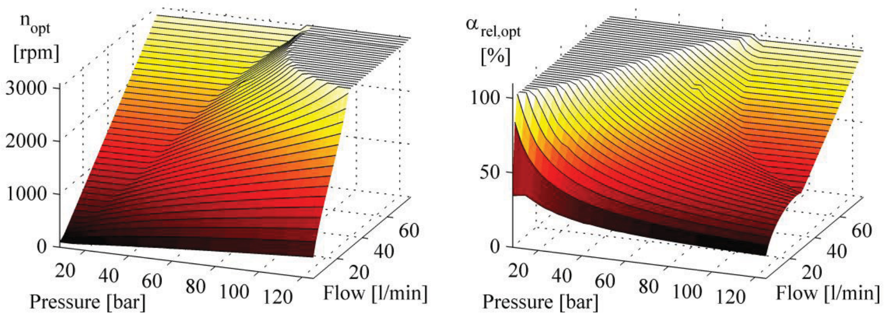

Figure 2: Energy optimized drive speed and pump displacement for the operation range of an exemplary HydroGear system

In terms of process control these maps can be stored inside the controller to track both actuating values. By doing so, in investigated applications energy savings of 20% and more could be achieved in comparison to pure speed variable or pure displacement variable pumps.

2.2. Model Predicitve Energy Optimization

In high dynamic process cycles the conventional energy optimization can be improved by considering additional system losses. Fast acceleration sequences of the electric drive lead to high motor torques and finally to increased current losses of the frequency converter as well as increased copper losses in the motor. In addition, due to cost reasons, most frequency converters have no energy recuperation capability. Therefore, the kinetic energy that is saved in the system inertia is lost during deceleration.

$$W_{kin} = \frac{1}{2} J_{tot} \cdot \frac{\pi}{30} \cdot (n_2^2 - n_1^2) \tag{7}$$

On the hydraulic side, the adjustment of the pump displacement requires a certain fluid volume that is taken from the high pressure side of the pump outlet. The resulting energy losses depend on the system pressure and the geometry of the swashplate actuator and can be interpreted as an additional dynamic leakage.

$$W_{Vol,dyn} \sim p \cdot \Delta\alpha \tag{8}$$

To consider the identified dynamic losses in the objective function of equation (6), the optimization of a single operation point has to be replaced by a holistic optimization of an entire process cycle. By means of a time discretization, a model predictive approach is presented in /8/ that minimizes the integral of the system losses for recurring processes.

$$min\{W_{tot}\} = min\left\{\int_0^{T_{Cyc}} P_{tot}(t) \cdot dt\right\} \xrightarrow{Discretization} min\left\{T \cdot \sum_{t=k\cdot T} P_{tot}(n_{k\cdot T})\right\} \tag{9}$$

As the dynamic optimization is based on the information of the time series of pressure $p(t)$ and volume flow $Q(t)$, the underlying hydraulic system can be treated as a black box. The model predictive approach, thus, can be applied to various actuators as hydraulic cylinders, hydraulic motors or any valve controlled subsystem. As a result the machine operator gets the optimum sequence of the drive speed and pump displacement that maximizes the energy efficiency. **Figure 3** shows the introduced concept.

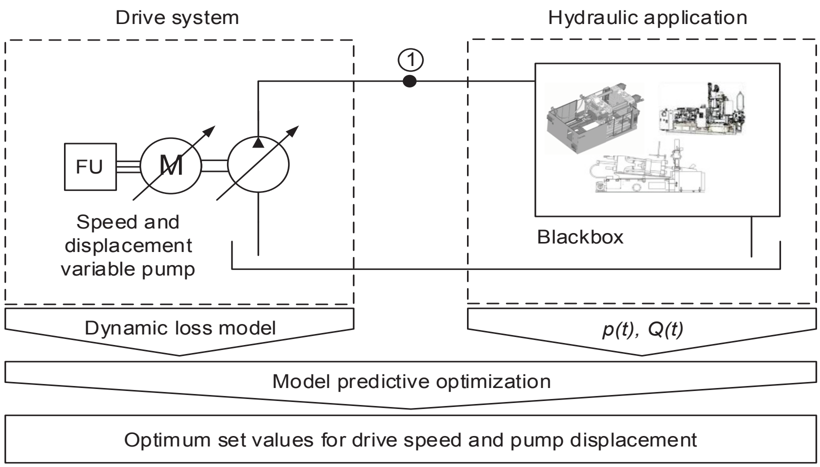

Figure 3: Model predictive energy optimization of HydroGear systems

2.3. Application Example

To evaluate the efficiency of HydroGear systems a stop-and-go movement of a hydraulic cylinder according to **Figure 4** will be discussed in this section.

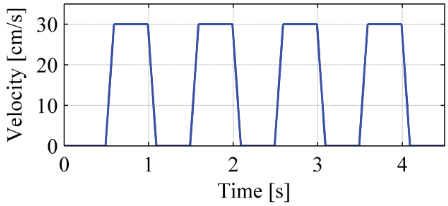

Figure 4: Application Example: Stop-and-go movement of a hydraulic cylinder

To apply the efficiency optimization, the investigated linear movement can be transformed into equivalent sequences of system pressure and volume flow. **Figure 5** shows the resulting sequences of drive speed and pump displacement for both concepts as well as for a pure displacement variable pump operated at a constant speed level as reference system (VDP).

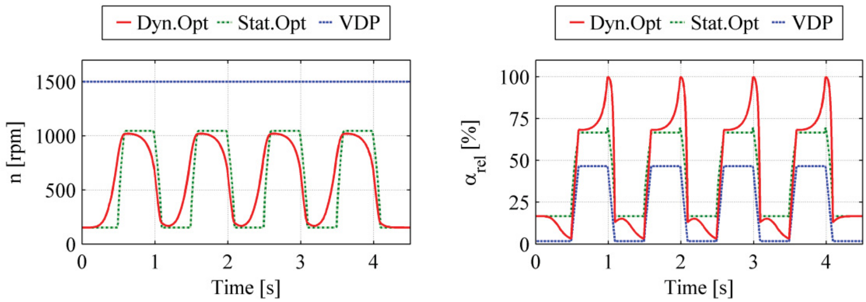

Figure 5: Drive speed and pump displacement for conventional optimization, model predictive optimization and a pure displacement variable pump

The usage of the drive speed as a second actuating variable in HydroGear systems is obvious in the left diagram. Regarding the conventional static optimization, mainly two constant levels for drive speed and pump displacement can be seen that correspond to the stop and the go phases of the movement. Looking at the dynamically optimized sequences, the model predictive approach avoids fast acceleration and deceleration of the electric drive. Instead, the demanded velocity profile of **Figure 4** is ensured by properly adjusting the pump displacement. For example an increase of the pump displacement allows a prior reduction of the drive speed and, thus, a reduction of the overall dynamic losses. **Figure 6** shows a comparison of the energy consumption of the stop-and-go movement for all three operation concepts that is normalized to the pure displacement variable pump running at constant speed (VDP).

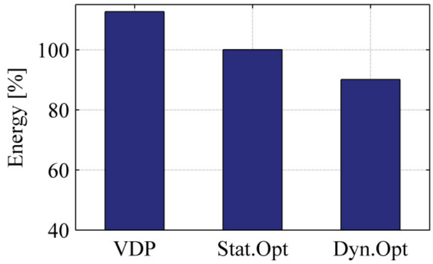

Figure 6: Energy Consumption of the compared control strategies

Both optimization concepts demonstrate the potential energy savings of HydroGear systems. While the conventional static optimization strategy can save 13%, the model predictive approach reaches further 10% energy savings by considering dynamic losses of the actuator systems.

3. Improving Process Dynamics with HydroGear Systems

Besides energy efficiency the degree of freedom of HydroGear systems can be used to maximize the reachable process dynamic. By means of a setpoint jump of the volume flow $Q_1 \rightarrow Q_2$ the potential of using two actuators simultaneously will be shown in the following sections.

3.1. State of the Art

At low volume flows HydroGear systems are usually operated at low drive speed levels due to energy reasons (cmp. **Figure 2**). In the case of a following setpoint jump, however, higher speed levels promises better performance as the effect of a displacement change $d\alpha/dt$ on the volume flow gradient dQ/dt increases proportional to the drive speed level n_0.

$$Q = n_0 \cdot V_0 \cdot \alpha \quad \rightarrow \quad \frac{dQ}{dt} = n_0 \cdot V_0 \cdot \frac{d\alpha}{dt} \tag{10}$$

Figure 7 illustrates this relation and shows a decreased rising time of the setpoint jump by increasing the drive speed.

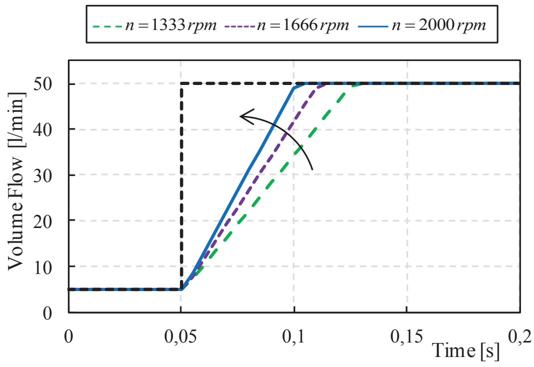

Figure 7: Influence of the speed level on the volume flow gradient of a speed and displacement variable pump

In former publications as well as industrial applications, thus, a prior increase of the drive speed is proposed. In this way, the subsequent setpoint jump can be operated at maximum drive speed while using the maximum dynamic of the displacement actuator at the same time /5/.

3.2. Process Adapted Control

Although the established strategy promises an excellent process dynamic it excludes the acceleration potential of the electric drive to optimize the setpoint jump. In /9/ a process adapted control concept has been proposed that minimizes the rising time by intelligently using the degree of freedom of HydroGear systems. In case of a synchronous use of both actuators equation (10) has to be expanded to:

$$Q = n \cdot V_0 \cdot \alpha \quad \rightarrow \quad \frac{dQ}{dt} = \frac{dn}{dt} \cdot V_0 \cdot \alpha + n \cdot V_0 \cdot \frac{d\alpha}{dt} \tag{11}$$

In combination with the equations of motion of drive speed $n(t)$ and pump displacement $\alpha(t)$ a non-linear equation system can be achieved and transformed into a constrained mathematical optimization problem.

$$n(t) = n_1 + \frac{1}{J_{\mathrm{tot}}} \int_{t_1}^{t} \left(M_{\max} - \frac{p \cdot \alpha(t) \cdot V_0}{2\pi} \right) \cdot d\tau \tag{12}$$

$$\alpha(t) = \alpha_1 + \int_{t_1}^{t} \dot{\alpha}(\tau) \cdot d\tau \tag{13}$$

The solution of this minimization task is the optimum initial drive speed n_1 to serve the initial volume flow $Q_1 = n_1 \cdot \alpha_1 \cdot V_0$. By accelerating with the maximum available motor torque $M_{\max}$ and swiveling out the pump's swash plate with the maximum possible gradient $\dot{\alpha}_{\max}$ the final volume flow $Q_2 = n_2 \cdot \alpha_2 \cdot V_0$ will be reached in the shortest possible time. **Figure 8** shows a descriptive explanation of the necessity to calculate the optimum initial drive speed.

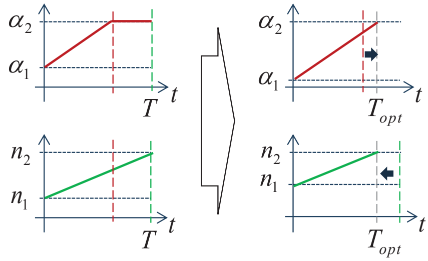

Figure 8: Simultaneous increase of drive speed and pump displacement

While on the left side the final values of drive speed and pump displacement are reached one after the other, an adjustment of n_1 on the right side ensures the simultaneous increase of both actuators and leads finally to the minimum rising time T_{opt}. A detailed derivation can be found in /9/.

With respect to the complexity of equations (12, 13) it gets clear that the known strategy of a simple prior increase to the maximum drive speed cannot be the optimum solution. **Figure 9** confirms this conclusion by adding the result of the dynamic maximizing strategy (DMS) on the left side.

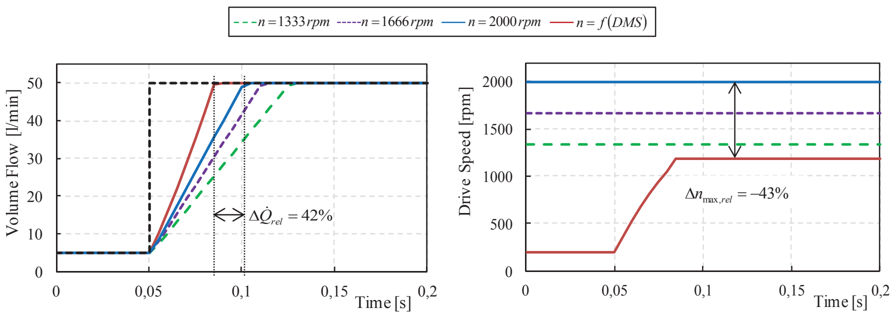

Figure 9: Dynamic maximizing strategy (DMS) of HydroGear systems

The optimized sequence of the drive speed on the right side shows an unexpected result. Instead of the known increase to the maximum drive speed the DMS proposes a decrease to only $n_1 = 200rpm$ to minimize the rising time in this case. By means of a following acceleration of the electric drive the volume flow gradient can be increased by over 40%. At the same time the maximum necessary speed level is almost bisected what proves that the pump has not to be operated at critical high drive speeds to realize high volume flow gradients. Besides the optimum process dynamic the DMS, thus, can contribute to a reduction of friction, wear and cavitation in the hydraulic pump, too. As a matter of course the benefit of the DMS depends on the dynamics of both actuators. A high performance synchronous motor will dominate a slow displacement actuator and vice versa. In practice, however, the performances of pump and motor are often in the same range.

4. Improving Noise Level with HydroGear Systems

Ever stricter legal requirements demand the machine manufacturers to reduce the noise level in their production machines. Speed variable pump drives allow to reduce the rotational speed at partial load and pressure holding sequences what leads to a clear reduction of the drive system’s noise level. Besides this obvious potential HydroGear systems offer the possibility to avoid defined speed ranges by efficiently choosing the operation point of the electric drive and the hydraulic pump. **Figure 10** shows the original drive speed histogram of a speed controlled process cycle on the left side.

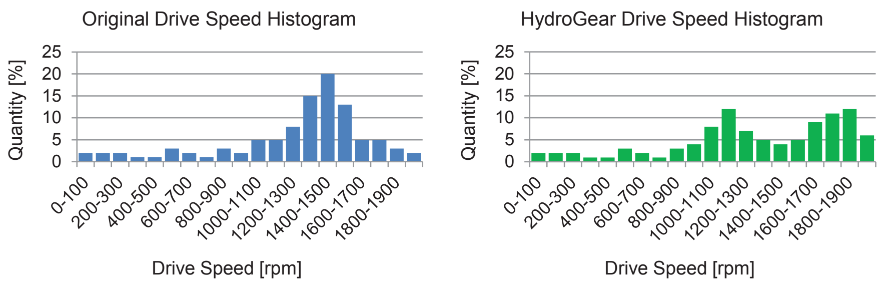

Figure 10: Suppression of critical resonance frequency in a HydroGear system

Assuming a noisy resonance frequency at a speed level of $\mathrm{n} = 1500rpm$ the HydroGear system suppresses efficiently the critical interval on the right side without effecting the hydraulic process itself. If necessary it is possible to avoid multiple speed ranges, too. In combination with the model predictive energy optimization of section 2.2, a weighted objective function can be defined that allows the machine operator to adjust the process control to the specific requirements of the application.

$$\min C = min\left\{\lambda \cdot \int_0^{T_{Cyc}} f_{losses}(t) \cdot dt + \mu \cdot \int_0^{T_{Cyc}} g_{noise}(t) \cdot dt\right\} \tag{14}$$

The functions $f_{losses}(t)$ and $g_{noise}(t)$ thereby represent appropriate mathematical descriptions of the overall system losses and the noise generation of the electro hydraulic drive system.

5. Serving Power Peaks with HydroGear Systems

Many process cycles have short power peaks that have to be considered for sizing of the drive system. In terms of speed variable pumps this includes the power supply, the frequency converter, the motor and the pump. To avoid oversizing of these components different energy storages can be integrated in the machine concept. Besides hydraulic accumulators and electric capacitors, the use of rotating masses as kinetic energy storage is widespread in industrial applications. **Figure 11** shows a DC coupling of an electro-hydraulic drive unit with a kinetic buffer motor that is used to serve short power peaks.

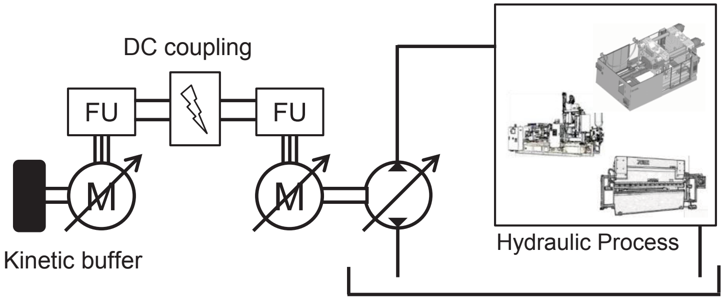

Figure 11: DC coupling of an electro hydraulic drive unit with a kinetic buffer motor

By adding further drive units to the same DC link an efficient power management system can be built to smooth the overall main power. As recuperated energy from a single drive unit can directly be used to load the kinetic buffer or serve another power demanding drive unit, additional hardware as braking resistors or more expensive regenerative frequency converters can be avoided. However, in a DC coupled network the energy exchange depends on the efficiency of all involved energy converters. Therefore, it is desirable to store the energy as close as possible to the drive unit that requires a power peak.

Looking at an electro hydraulic drive system the rotating inertia of motor, pump and coupling is in principle a kinetic energy storage. As the drive speed is directly coupled to the volume flow of the pump, however, loading and unloading the internal kinetic buffer affects the process control. At this point HydroGear systems offer the possibility to compensate the speed variation by means of the variable pump displacement. The degree of freedom allows to decouple the drive speed and the volume flow and finally to use the internal inertia to serve short power peaks without the need of any additional hardware. **Figure 12** illustrates the concept with a short power peak occurring at $\mathrm{t} = 1.5\mathrm{s}$.

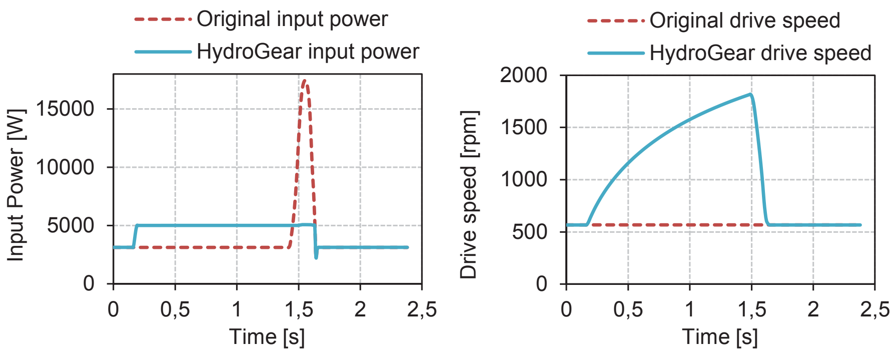

Figure 12: Serving power peaks with HydroGear systems

By means of a prior increase of the drive speed (right) and a simultaneous reduction of the pump displacement, the kinetic buffer can be loaded without affecting the hydraulic process. When the power peak occurs the drive gets strongly decelerated to release the stored kinetic energy. As a result the maximum electrical main power can be noticeably reduced. Instead a small power offset is visible during the loading phase of the kinetic buffer in **Figure 12**. The possible power reduction depends on the given process as the volume flow demand during the power peak defines the minimum possible speed level n_{min}. The storable energy is finally given by equation (15).

$$n_{min} = \frac{Q_{Peak}}{V_0} \quad \rightarrow \quad \mathrm{W}_{kin} = \frac{1}{2} \cdot J_{tot} \cdot \left(n_{max}^2 - n_{min}^2\right) \tag{15}$$

As this energy is mechanically stored in the pump shaft, HydroGear systems allow a potential downsizing of power supply, frequency converter and electric drive.

6. Summary and Outlook

HydroGear systems combine the advantages of frequency controlled and displacement controlled pumps. The two set values offer a degree of freedom for process control that can efficiently be used to increase the performance of electro hydraulic drive systems. By means of a dynamic loss model of the drive system the energy consumption can be reduced by 20% and more comparing to pure speed or pure displacement variable concepts. Considering process dynamics it was shown that a simultaneous use of the two actuators ensures the best possible performance. The volume flow gradient could exemplarily be improved by 40%. Besides energy efficiency and process dynamics, an improved noise level and a reduction of electrical power peaks are benefits of Hydro-Gear systems. To maximize the impact intelligent control algorithms are necessary. To follow multiple objectives at once, future studies will focus on efficient combinations of the presented concepts.

7. References

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8. Nomenclature

α	Swivel Angle	%
I	Motor Current	A
J_{tot}	Inertia of drive system	kgm^2
$K_{f,1/2}$	Drive Constant: Iron Losses – Electric Drive	Ws, Ws^2
K_{FC}	Drive Constant: Current Losses – Frequency Converter	W/A
M_{max}	Maximum Motor Torque	Nm
M_R	Pump Friction Torque	Nm
n	Drive Speed	rpm
p	Pressure	bar
P_{Cu}	Copper Losses in the Synchronous Motor	W
P_{FC}	Current Losses in the Frequency Converter	W
P_{Fe}	Iron Losses in the Synchronous Motor	W
P_{Mech}	Mechanical Losses in the Pump	W
P_{tot}	Overall System Losses	W
P_{Vol}	Volumetric Losses in the Pump	W
Q	Volume Flow	l/min
Q_L	Pump Leakage	l/min
R_{Cu}	Winding Resistance – Electric Drive	Ohm
T_{cyc}	Cycle Time	sec
V_0	Pump Displacement	ccm
W_{kin}	Kinetic Energy	J
W_{tot}	Overall Energy Consumption	J